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TECHNICAL NOTE 2799

SIMPLE GRAPHICAL SOLUTION OF HEAT TRANSFER AND EVAPORATION FROM SURFACE HEATED TO PREVENT ICING

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SUMMARY

Equations expressing the heat transfer and evaporation from wetted surfaces during ice prevention have been simplified and regrouped to permit solutions by simple graphical means. Working charts for quick and accurate anti-icing calculations are also included.

INTRODUCTION

Solution of the general problem of heat and mass transfer from a wetted surface in forced convection is quite involved and tedious. The calculations that often must be made point by point along a surface are complex even after the basic variables and ambient conditions have been determined. The method of solution customarily performed involves several trial-and-error calculations that are intermediate between the final answer and the basic factors that define a particular anti-icing situation.

This investigation was conducted at the NACA Lewis laboratory to simplify the method of calculating heat and mass transfer after the basic factors (such as ambient temperature, rate of water interception, and relative velocity) are known. Calculations will be greatly simplified by a fundamental rearrangement of terms in the conventional equations, which will eliminate the trial-and-error calculations and permit a rapid graphical solution. The solution will be limited to the normal range of anti-icing conditions and to the case of surface temperatures above 32° F with liquid-water interception by the surface.

DEVELOPMENT OF GRAPHICAL SOLUTION

Simplification of Conventional Equations

The solutions most frequently desired in anti-icing calculations are for the heat-transfer rate associated with prescribed anti-icing conditions and the corresponding evaporation rate. These terms constitute the two principal solutions of the anti-icing problem and are developed separately.

Heat transfer. - The heat transfer in forced convection from a surface subjected to water impingement and heated above 32° F has been represented by the total heat transferred by convection and by evaporation of surface water, plus the sensible heat change of the impinging water. These heat-transfer processes are discussed in detail in references 1 to 2332
5 and are summarized in the following five conventional equations:

$$q = h_a(t_s - t_d) + h_a(X - 1)\ K\ (t_s - t_d) + M_w c_w (t_s - t_w) \tag{1}$$

where

$$t_d = t_0 + \frac{V_0^2}{2gJc_p}\left[1 - \left(\frac{V_l}{V_0}\right)^2 (1 - R)\right] - 0.622\ \frac{LK}{c_p}\left(\frac{e_d - e_l}{p_l}\right) \tag{2}$$

$$X = 1 + \left(\frac{e_s - e_d}{t_s - t_d}\right)\left(\frac{0.622L}{p_l c_p}\right) \tag{3}$$

$$t_w = t_0 + \frac{V_0^2}{2gJc_w} \tag{4}$$

and

$$\log_{10}\left(\frac{p_l}{p_0}\right) = \left(3.49 + 9.05\ \frac{e_0}{p_0} + 0.000683\ \frac{m_0 T_0}{p_0}\right) \log_{10}\left(\frac{T_l}{T_0}\right) + 3.93\left(\frac{L_l e_l}{T_l p_l} - \frac{L_0 e_0}{T_0 p_0}\right) \tag{5}$$

(The symbols used herein are given in the appendix.)

A solution of equation (1) requires the prior calculation of equations (2) to (5). Because of the dependence of e_d on t_d and of e_l on t_l, equations (2) and (5) are both trial-and-error solutions. In order to eliminate these laborious calculations, a single equation will be derived containing only a minimum number of basic variables.

The number of basic variables that define an anti-icing situation can be conveniently reduced to five: t_s, t_0, V_0, p_0, and p_l. In addition, three complex terms that cannot at the present time be generally reduced to basic variables must be retained in the final expression.

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These terms h_a, M_w, and K are discussed in a subsequent section of this report. In order to obtain a simplified relation using the preceeding eight terms, the following assumptions are made:

(1) In the air flow over a body, temperatures and pressures change adiabatically too quickly to permit any change in the state of entrained moisture; the local-stream vapor pressure (as shown in reference 2) is then given by

$$e_l = e_0 \frac{p_l}{p_0} \tag{6}$$

(2) The local velocity and pressure may be related by the incompressible dry-air relation

$$V_l^2 = V_0^2 - \frac{2}{\rho_0}(p_l - p_0)\ 70.7 \tag{7}$$

Further simplification is obtained by expressing the following minor variables as constants near the middle of their expected range in anti-icing calculations:

(1) Recovery factor R, assumed as 0.85, corresponding to the conservative case of laminar flow

(2) Latent heat of vaporization L, assumed as 1066 Btu per pound, corresponding to an average surface temperature of 50° F

(3) Ambient absolute temperature (for air density calculation), assumed as 464° R

With the use of these assumptions and arbitrary constants, equations (1) to (7) may be combined as follows:

$$\frac{q}{h_a} = t_s - t_0 - \frac{V_0^2}{1.2\times10^4} + \frac{0.15}{1.2\times10^4}\left[V_0^2 - \frac{2\ (464)(p_l - p_0)\ 70.7}{0.0412\ p_0}\right]$$

$$+ \frac{0.622\ K\ (1066)}{0.24}\left(\frac{e_s - e_l}{p_l}\right) + \frac{c_w M_w}{h_a}\left(t_s - t_0 - \frac{V_0^2}{5\times10^4 c_w}\right) \tag{8}$$

from which

$$\frac{q}{h_a} = (t_s - t_0)\left(1 + \frac{M_w c_w}{h_a}\right) - \frac{V_0^2}{5\times10^4}\left(3.55 + \frac{M_w}{h_a}\right) + 2760\ K\left(\frac{e_s}{p_l} - \frac{e_0}{p_0}\right) + 19.9\left(1 - \frac{p_l}{p_0}\right) \tag{9}$$

The term q/h_a of equation (9) is measured dimensionally in degrees of temperature, and is now expressed as the arithmetic sum of five independent terms, as follows:

$$\frac{q}{h_a} = \tau_1 - \tau_2 + K(\tau_3 - \tau_4) + \tau_5 \tag{10}$$

where from inspection of equation (9),

$$\tau_1 = (t_s - t_0)\left(1 + \frac{M_w c_w}{h_a}\right) \tag{11}$$

$$\tau_2 = \frac{V_0^2}{5 \times 10^4}\left(3.55 + \frac{M_w}{h_a}\right) \tag{12}$$

$$\tau_3 = 2760\left(\frac{e_s}{p_l}\right) \tag{13}$$

$$\tau_4 = 2760\left(\frac{e_0}{p_0}\right) \tag{14}$$

$$\tau_5 = 19.9\left(1 - \frac{p_l}{p_0}\right) \tag{15}$$

These τ terms are in a form that permits graphical presentation, as will be shown in the section Graphical Solution.

Mass transfer. - The rate of evaporation from a wetted surface above 32° F is given in reference 2 as

$$M_{ev} = 0.622\, h_a\, K\left(\frac{e_s - e_l}{p_l c_p}\right) \tag{16}$$

When equation (6) is substituted in the preceding equation,

$$M_{ev} = 0.622\, \frac{h_a K}{c_p}\left(\frac{e_s}{p_l} - \frac{e_0}{p_0}\right) \tag{17}$$

or

$$M_{ev} = \frac{h_a K}{1066}(\tau_3 - \tau_4) \tag{18}$$

When the surface is dry (no impingement or run back), K and M_w are zero and equation (10) simplifies to

$$\frac{q}{h_a} = \tau_1 - \tau_2 + \tau_5 \tag{19}$$

When the rate of impingement M_w is lower than the rate of evaporation given by equation (18), the surface remains essentially dry because the water evaporates as it impinges, and the following equation applies:

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$$\frac{q}{h_a} = \tau_1 - \tau_2 + \tau_5 + 1066\,\frac{M_w}{h_a} \tag{20}$$

Equation (20) results from the addition to equation (19) of a term which includes the heat required to evaporate the impinging water completely.

Graphical Solution

<u>General solutions</u>. - The general equations of heat and mass transfer from a surface subjected to water impingement and heated above freezing (equations (10), (18), and (20)) have been formulated using the five terms τ_1 to τ_5. These τ terms are presented graphically in figure 1 and are determined by intersections, as follows:

	Abscissa		Sloping line
τ_1 = Intersection	M_w/h_a	and	$t_s - t_0$
τ_2 = Intersection	M_w/h_a	and	V_0
τ_3 = Intersection	p_l	and	t_s
τ_4 = Intersection	p_0	and	t_0
τ_5 = Intersection	p_l	and	p_0

A numerical example is shown by arrows and dashed lines on figure 1 to illustrate the use of the graph for the two principal solutions. The calculations required for the two solutions (using equations (10) and (18)) are quite simple after the eight initial variables are given.

Figure 1 may be used in conjunction with equations (10) and (18) to solve for any of the factors appearing therein, provided only one factor is unknown. Solutions may be made directly for the factors q, M_{ev}, K, and V_0, whereas the other factors, h_a, p_0, p_l, t_s, and t_0 each appear twice and must be obtained by trial-and-error. These trial-and-error solutions converge rapidly because the calculations required are minor and trends can be visualized from the graphs in figure 1. When q is zero, a solution for t_s yields the unheated datum, or equilibrium, surface temperature.

The coordinate scales used in the working chart of figure 1 should be large enough for the usual requirements of accuracy, permitting anti-icing calculations to be made quickly from the graph alone. In addition, the graph illustrates the relative magnitude of the various component terms and facilitates comparisons, check calculations, and evaluation of assumptions and short-cuts.

The ambient temperature, pressure, and velocity terms in equation (10) are determined for the undisturbed stream conditions just ahead of the body in question. When the body is located in a pipe or duct, the ambient terms should be evaluated for conditions in the duct ahead of the body, and not for the external air-stream conditions.

Point and area solutions. - The solutions described previously have been based on local point values of the primary variables and expressed for unit surface areas. If appreciable variations occur in the surface-wise distribution of these primary factors, the local q and M_{ev} solutions, as previously determined, should be obtained for several points along the surface in the direction of flow starting at the stagnation region. The total heat and mass transfer from a specific surface area may then be obtained by graphical or numerical integration. In this point-by-point manner, abrupt changes in surface conditions will be taken into consideration, for example, (a) the decrease in the external heat-transfer coefficient immediately downstream of the stagnation region, (b) the downstream limit of water impingement, (c) the transition from laminar to turbulent heat-transfer, (d) the aft limit of run-back rivulets, and (e) any sudden changes in surface contour or surface temperature. Furthermore, in this manner it is possible to determine how far downstream on a surface the impinged water will run back before it is evaporated by summing up the cumulative impingement and evaporation from the stagnation region downstream to a particular point; thus

$$M_r = \sum_{S=0}^{S=S_l} (M_w - M_{ev})\,\Delta S \qquad (21)$$

When M_r becomes zero all the run back will have been evaporated.

Although the point-by-point solutions discussed are usually required for accuracy, the graph and associated equations are equally applicable to area solutions in which the primary factors are constant over a given area, or approximately constant and can be conveniently averaged. In such a case, area solutions are obtained in one operation by multiplying q and M_{ev} by the specific areas involved.

Solutions with gas heating. - With a heating source of hot gas, the local heating rate q is not as directly measurable as with electric heating; whereas the input wattage is known for an electric heater, with gas heating, gas temperature, flow rate, and passage design are usually known. For thin conductive skins between the gas supply and the external air stream and neglecting radiation, the following equation may be used:

$$q = h_g(t_g - t_s) \tag{22}$$

The q values of equations (22) and (10) may then be equated and solved for any one unknown. If, for example, surface temperature t_s is unknown, a trial solution may be made by estimating t_s and proceeding until a check is obtained on the q values of equations (22) and (10).

Usually an appreciable drop in gas temperature t_g occurs in the direction of flow, necessitating a stepwise analysis in the flow direction. Detailed treatments of surface evaporation and gas-heating considerations are given in references 1 and 4.

SUPPLEMENTARY GRAPHS

Of the eight factors and variables required for solution of equations (10) and (18) (with fig. 1), several are involved and tedious to calculate. Graphs that simplify these calculations have been included wherever possible to do so in general terms. In the usual icing or heat-transfer problem, the ambient conditions, the free-stream velocity, and the geometric-shape factors are known and constant for a given set of calculations; these known factors can sometimes be plotted to yield the required factors, which may be considered dependent variables.

Local Pressure and Velocity

The conversion between local static pressure p_l and local stream velocity v_l for the subsonic range of Mach number M is shown in figure 2 for compressible dry-air flow. Local velocity ratios may be directly converted into local pressure ratios after the free-stream Mach number curve has been located. The free-stream Mach number is determined from the relation

$$M_0 = \frac{V_0}{49.1\sqrt{T_0}} \tag{23}$$

Heat-Transfer Coefficients

External air flow. - In general, the value of the dry-air convective heat-transfer coefficient h_a is difficult to establish and several important assumptions are necessary. Heat-transfer coefficients are available for air flow over specific shapes, such as flat plates and cylinders (reference 6), over airfoils (references 1, 3, 7, and 8), and over bodies of revolution (references 9 and 10). The coefficients for flat plates and cylinders are, at present, general in application and are presented graphically in figure 3. In figure 3(a) the heat-transfer coefficients are given for laminar and turbulent air flow over flat plates and for the leading-edge stagnation point of cylinders. The ratio of local to stagnation point heat-transfer coefficient over the forward surface of a cylinder is shown in figure 3(b). The coefficients (fig. 3(a)) are determined by the use of an air-flow parameter consisting of air velocity, static pressure, and a significant length dimension. In order to simplify these relations, the absolute-temperature terms for surface and ambient air were taken as the same constants assumed for the derivation of equation (10). Errors resulting from this simplification are negligible when the many uncertain factors present in icing conditions are considered.

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The transition from laminar to turbulent heat-transfer is not easily located when the surface is covered with a water film or rivulets. In such cases, transition is very apt to be forward of the transition point located for flow over a dry surface. Limited evidence (reference 3) indicates that transition probably occurs near the downstream limit of droplet impingement on an airfoil. For unusual shapes and conditions, estimates are required for locating transition.

Internal gas flow. - The heat-transfer coefficient between a gas flowing in a confined passage and the wall of the passage is a complex function. The coefficient is known to vary with the distance from the entrance to the passage, with the passage shape, with position around the passage perimeter, with the gas-flow rate, and with the gas temperature (references 6, 11, and 12). The flow condition generally applicable to anti-icing calculations is that of fully developed turbulent flow in passages for which entrance length and position along the perimeter are assumed to have no effect. A condition of this type is shown in figure 4, in which the internal heat-transfer coefficient is obtained from the parameter $h_{g,t}(Ap/P^{0.2})$ plotted as a function of the gas flow and gas temperature. The term $(Ap/P^{0.2})$ varies only with passage geometry and will generally be constant for a series of calculations involving a specific flow passage. Figure 4 does not include radiant heat transfer.

Water-Interception Rate

Sufficient data are not available to permit generalization of the impingement term M_w for application to airfoils and bodies at various flight and ambient conditions. Impingement values therefore must be estimated on the basis of published data for specific airfoils (references 13 and 14) and for cylinders and spheres (reference 15) and from

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the knowledge of impingement limits learned from icing observations and photographs (references 3 and 16 to 20). Occasionally, assistance is obtained from the maximum condition of impingement that occurs with high airspeed, large droplet size, and small body size, for which straight line trajectories of water drops may be assumed. Under these conditions $M_w = (m/4.45)(V_0 \sin \gamma)$, where γ is the angle between the tangent to the surface at the point of impingement and the free-stream relative wind vector.

Surface Wetness

In the absence of exact data, the wetness fraction K in the basic equations is usually taken as 1 in the impingement area. Near the downstream limit of impingement, rivulets of run-back water commence and K decreases very rapidly. The K factors in regions of rivulet flow probably vary with many conditions, but limited data (reference 3) indicate that the value of K decreases in this region from a value near 0.3 down toward zero at the point of complete evaporation of run back.

CONCLUDING REMARKS

In the preceding derivations, the assumption is made that the moisture in the ambient air does not have time to change state during the adiabatic pressure and temperature changes in the local air stream. Because of the high speeds and short distances involved in most anti-icing calculations, this assumption appears reasonable. Answers obtained by use of this assumption do not differ appreciably from those using the assumption of phase changes in equilibrium with the local air temperature. The difference between the two solutions is greatest when p_l/p_0 is most removed from unity and when low heating rates are used.

Errors in the present graphical solution may become appreciable at high subsonic speeds because an incompressible-flow relation was used in eliminating the velocity-ratio term from the equations. Recovery factors for wetted surfaces at high speeds may also deviate from the value assumed herein. The errors in solution due to the temperature factors that were taken as constants near the mean of their expected ranges are in the order of 3 percent for the range of graph presented. The foregoing errors are negligible, however, compared with the uncertainty and the latitude involved in estimating values of the basic variables, especially the water-interception rate and the heat-transfer coefficient.

The form of solution presented herein is intentionally made general for maximum usage. The solution is applicable to practically any body shape for which the flow conditions are known, and is not dependent upon any specific assumptions as to body geometry, location of transition, wetness of surface, and method or extent of heating.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
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APPENDIX - SYMBOLS

The following symbols are used in this report:

A_p	cross-sectional area of duct or pipe passage, sq ft
c_p	specific heat of gas at constant pressure, 0.24 Btu/(lb)($^{\circ}$F)
c_w	specific heat of liquid water, 1.0 Btu/(lb)($^{\circ}$F)
D	diameter of cylinder, ft
e	partial pressure of water vapor, (corresponding to saturated air unless noted otherwise), in. Hg
g	acceleration due to gravity, 32.2 ft/sec^2
h	dry convective heat-transfer coefficient, Btu/(hr)(sq ft)($^{\circ}$F)
J	mechanical equivalent of heat, 778 (ft)(lb)/Btu
K	surface-wetness fraction
L	latent heat of vaporization of water, Btu/lb
M	Mach number
M_{ev}	rate of evaporation of water, lb/(hr)(sq ft)
M_r	rate of run back of surface water, lb/(hr)(ft span)
M_w	rate of interception of water, lb/(hr)(sq ft)
m	liquid-water content of ambient air, g/cu m
P	inside perimeter of duct or pipe, ft
p	absolute static pressure, in. Hg
q	rate of heat transfer, Btu/(hr)(sq ft)
R	kinetic energy recovery factor
S	surface distance from stagnation point in stream direction, ft
T	absolute temperature, $^{\circ}$R

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t temperature, $^{\circ}F$

V relative velocity between surface and stream, ft/sec

W rate of heated gas flow, lb/sec

w rate of heated gas flow, lb/hr

X Hardy's evaporation factor (equation (3))

γ angle of impingement of water droplet upon surface, deg

ρ density, $(lb)(sec^2)/ft^4$

τ_1 to τ_5 temperature terms defined by equations (11) to (15) and determined by intersections on figure 1, $^{\circ}F$

φ cylinder center-angle between stagnation point and point on cylinder surface, deg

Subscripts:

0 undisturbed ambient stream

a external air

d heat-transfer datum

g internal hot gas

l local

lam laminar flow

s surface

stag cylinder stagnation point

t turbulent flow

w water

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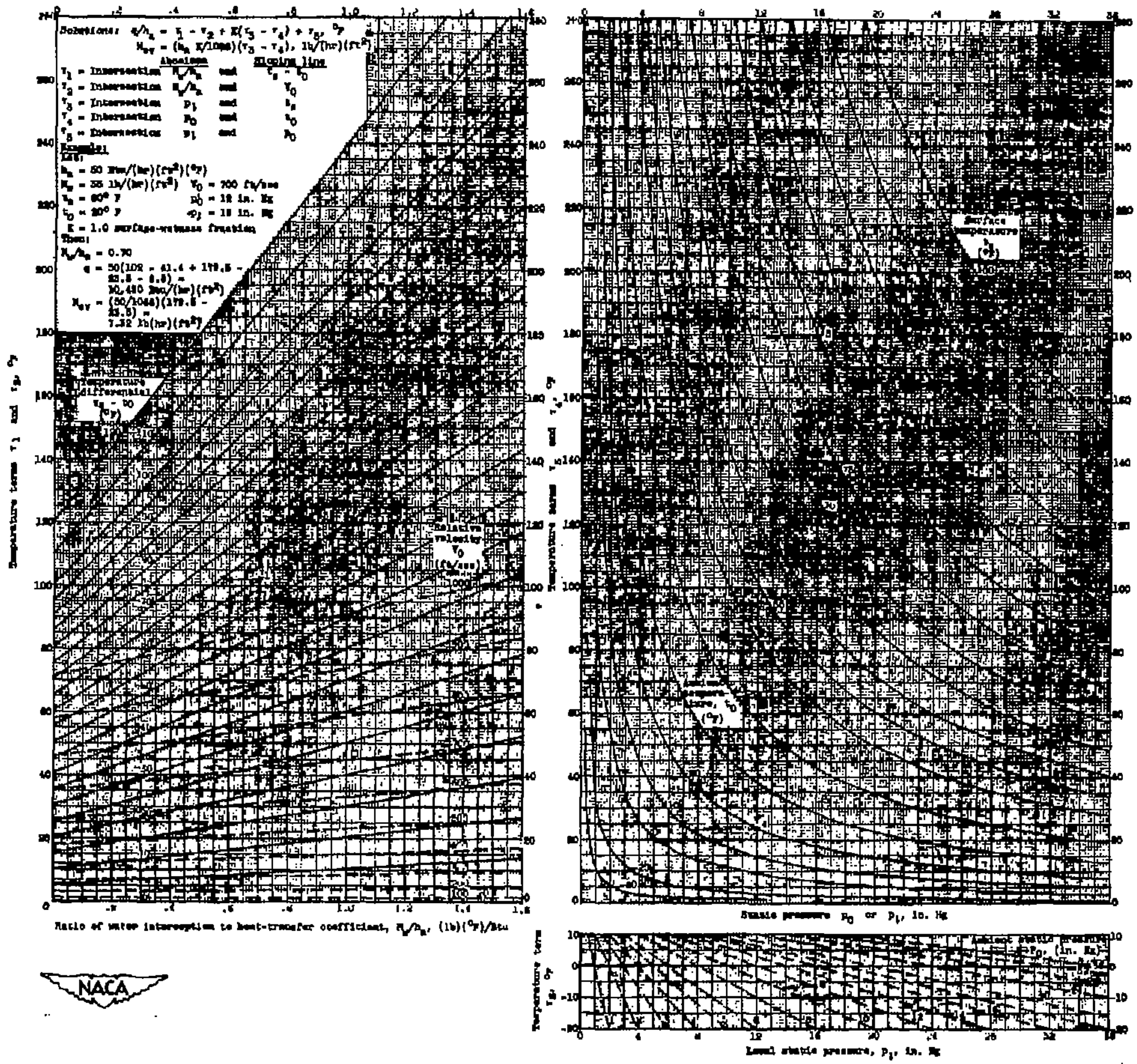


Figure 1. - Graphical solution of anti-icing heat and mass transfer from surface subject to impingement and heated above freezing. (A $17\frac{1}{2}\times22$ inch print of this figure is attached.)

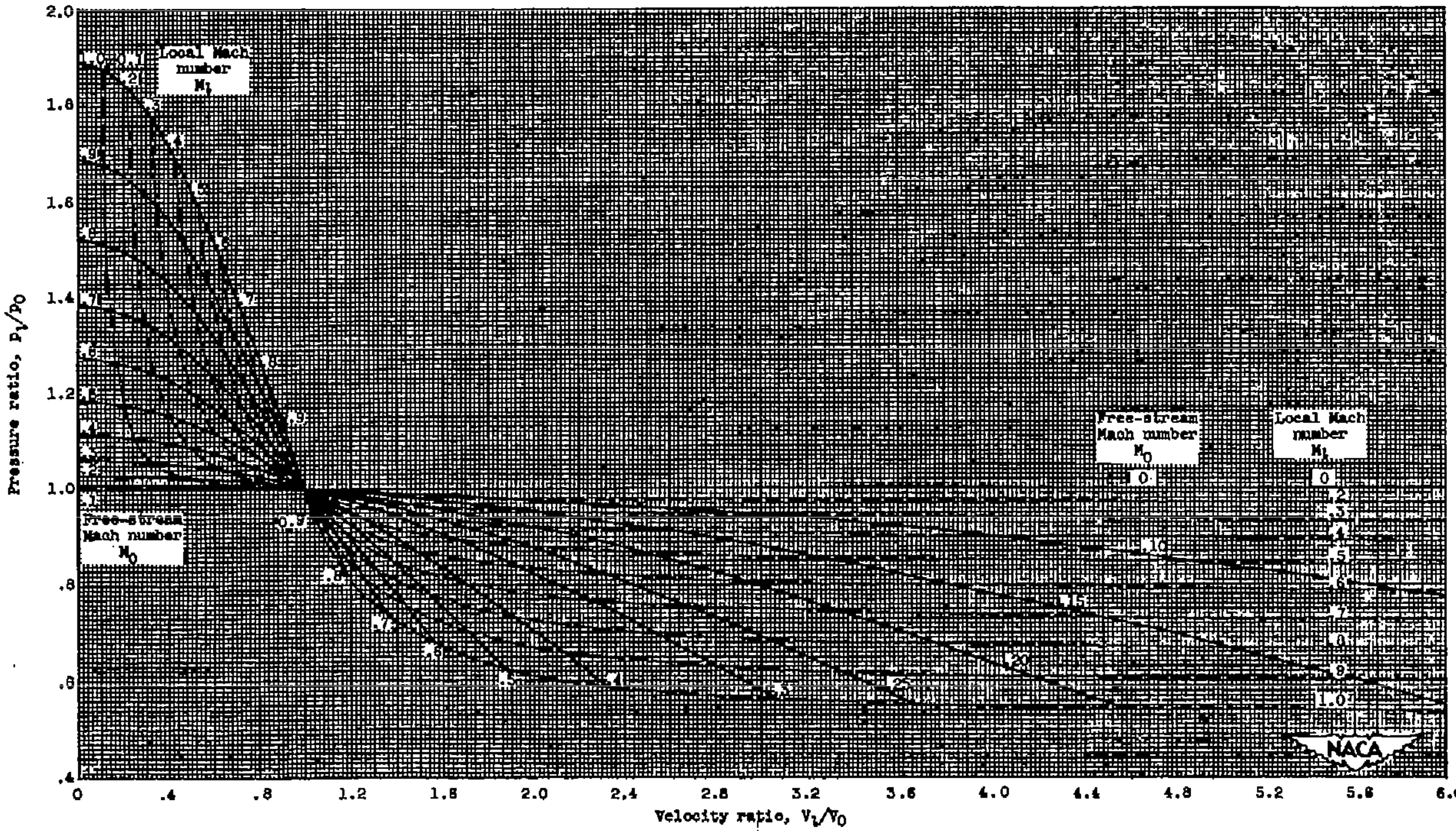
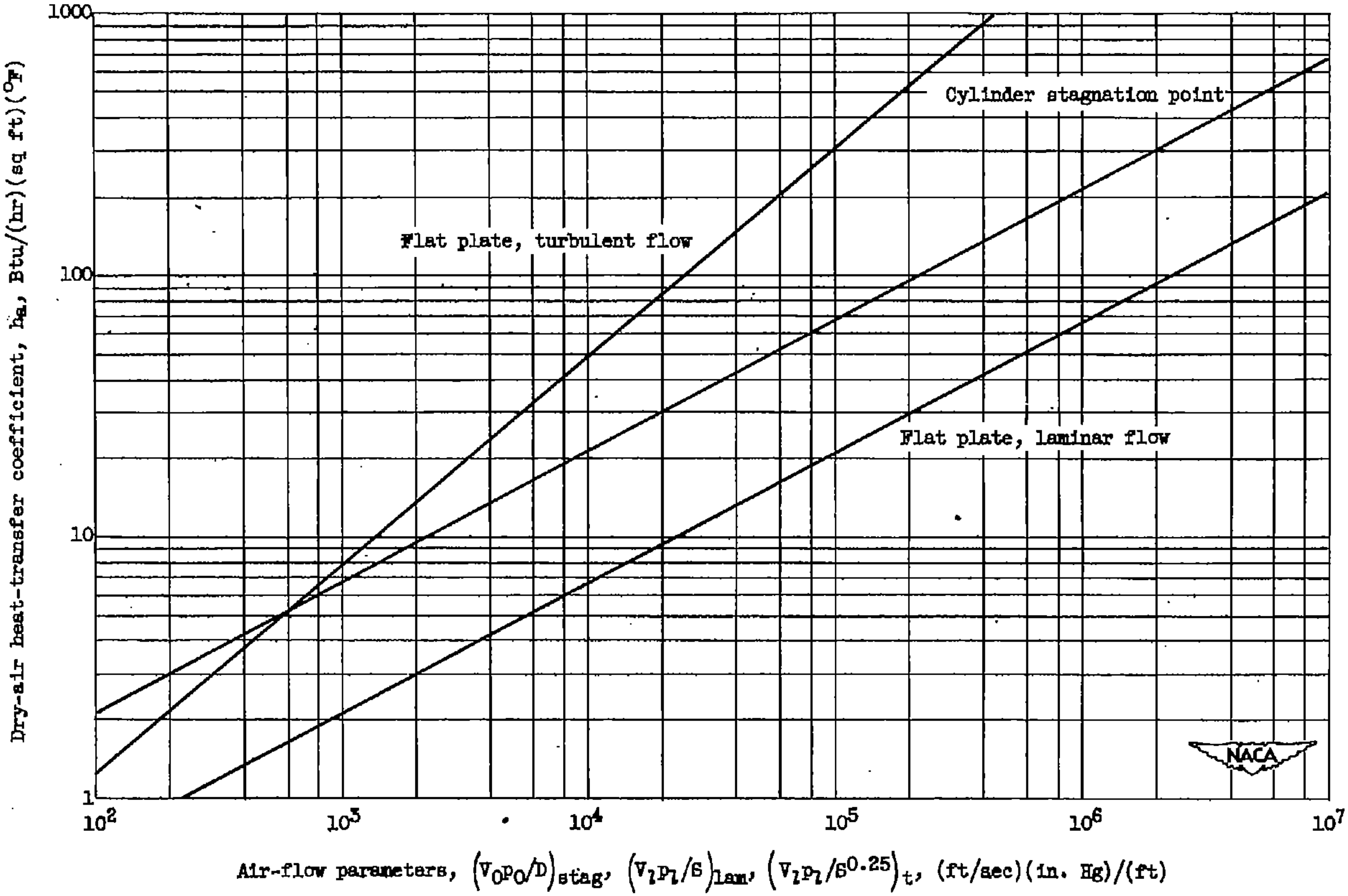
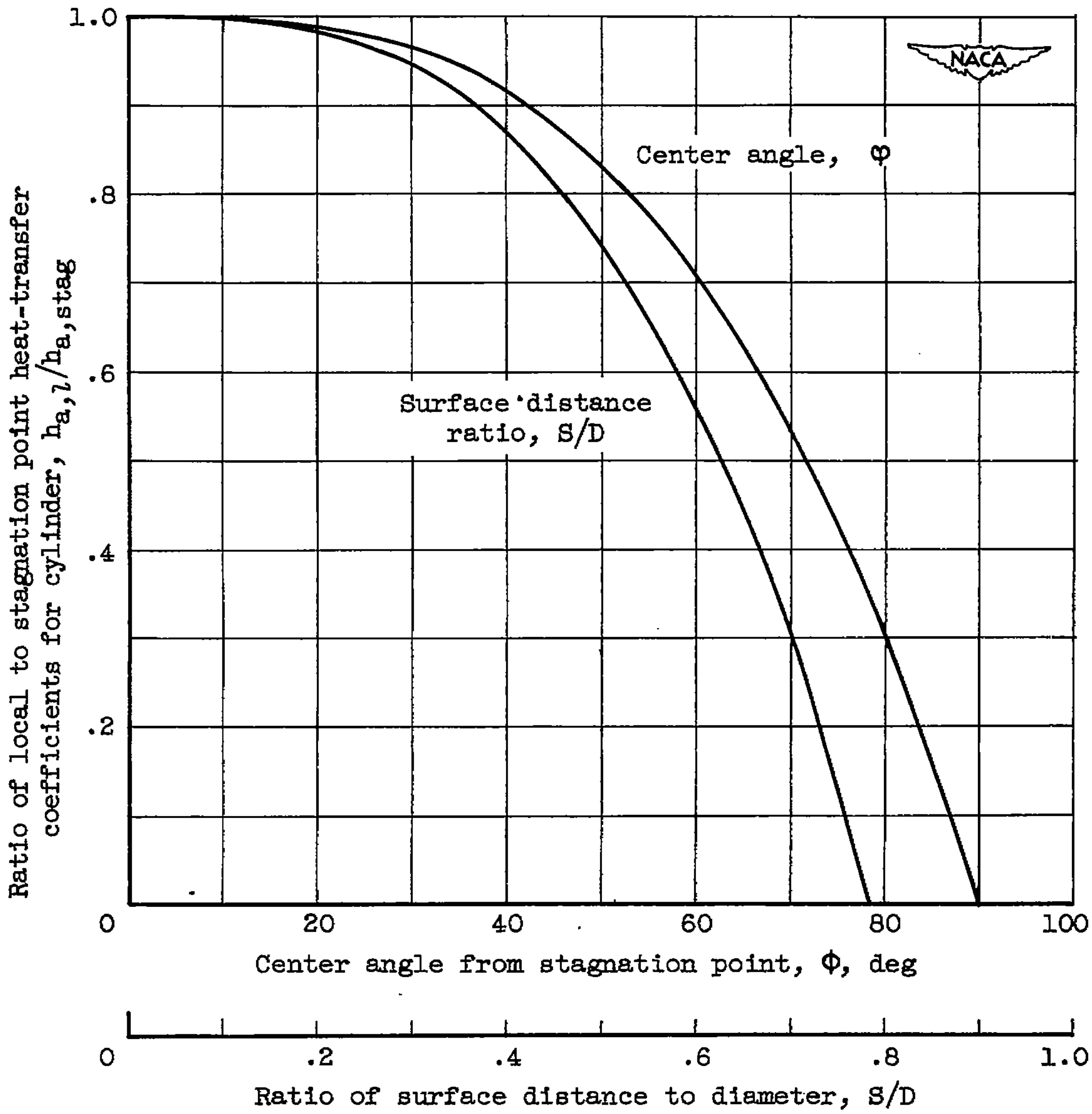


Figure 2. - Velocity and pressure ratios as function of free-stream and local Mach numbers for subsonic compressible flow.



(a) Dry-air heat-transfer coefficients for flat plates and cylinder stagnation points as functions of flow velocity, static pressure, and significant length.

Figure 3. - External heat-transfer coefficients.



(b) Variation of heat-transfer coefficient over forward surface of cylinder as function of coefficient at stagnation point.

Figure 3. - Concluded. External heat-transfer coefficients.

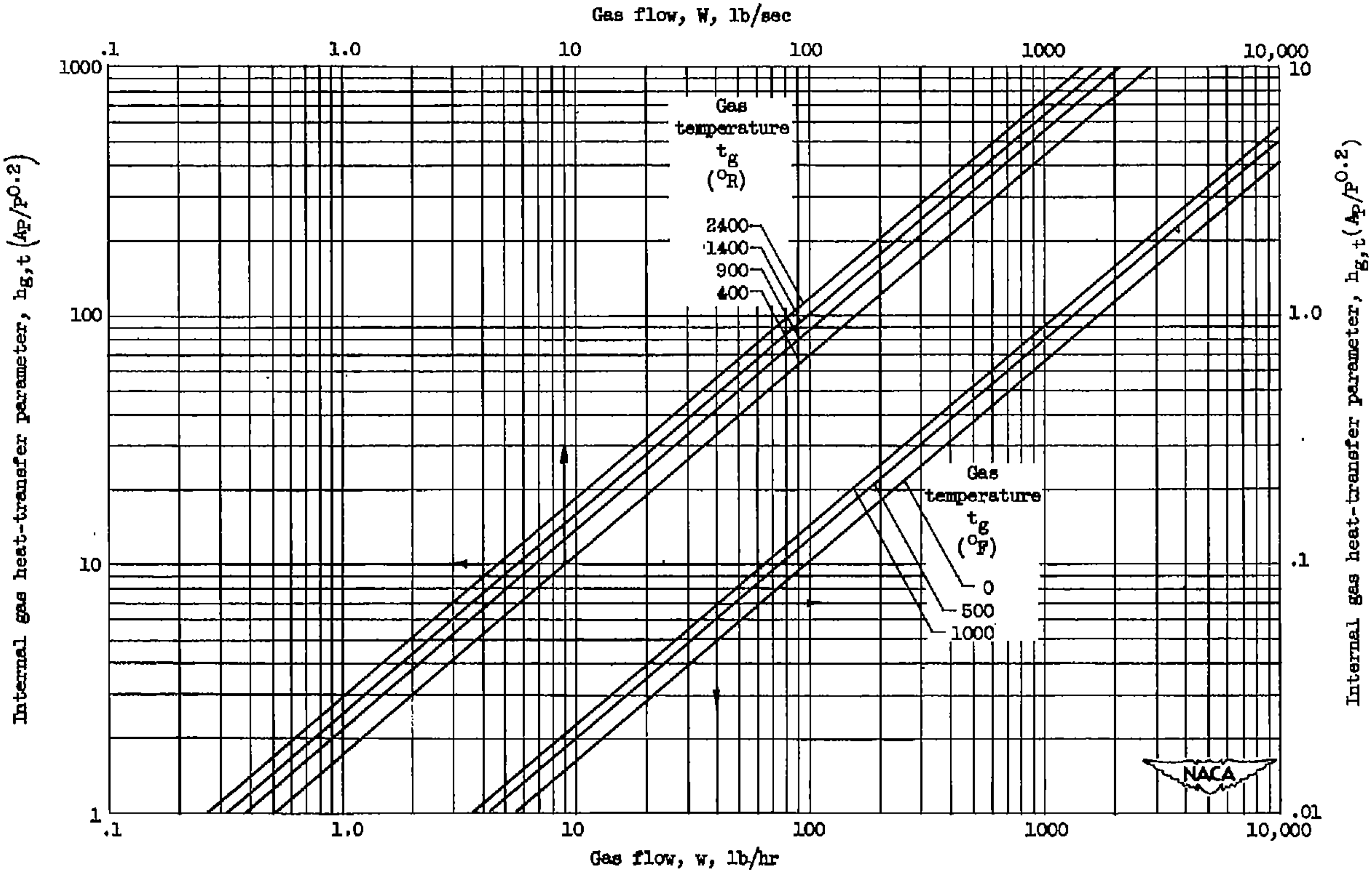


Figure 4. - Determination of turbulent pipe-flow heat-transfer coefficient in terms of gas flow, temperature, and passage size.

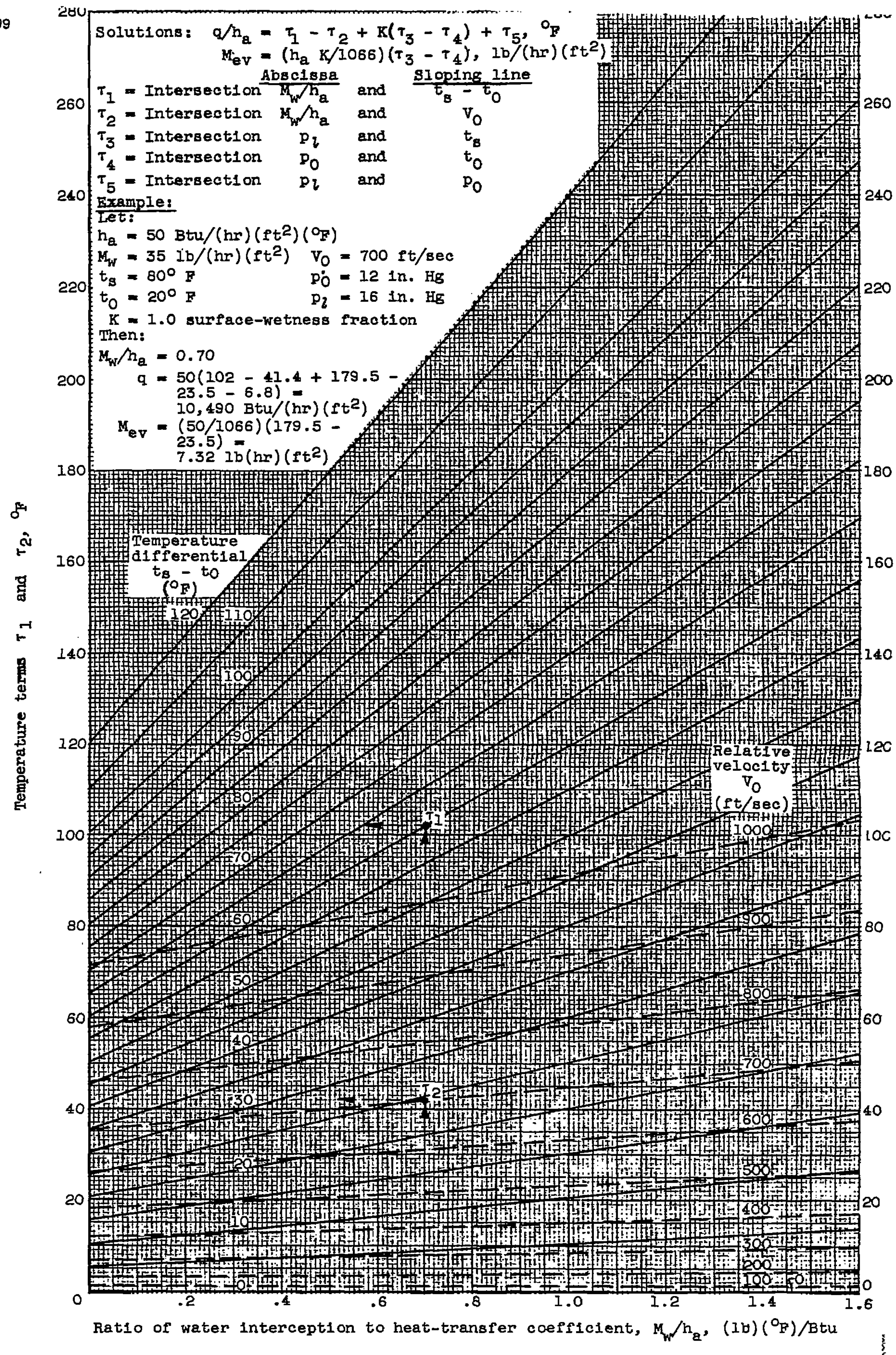
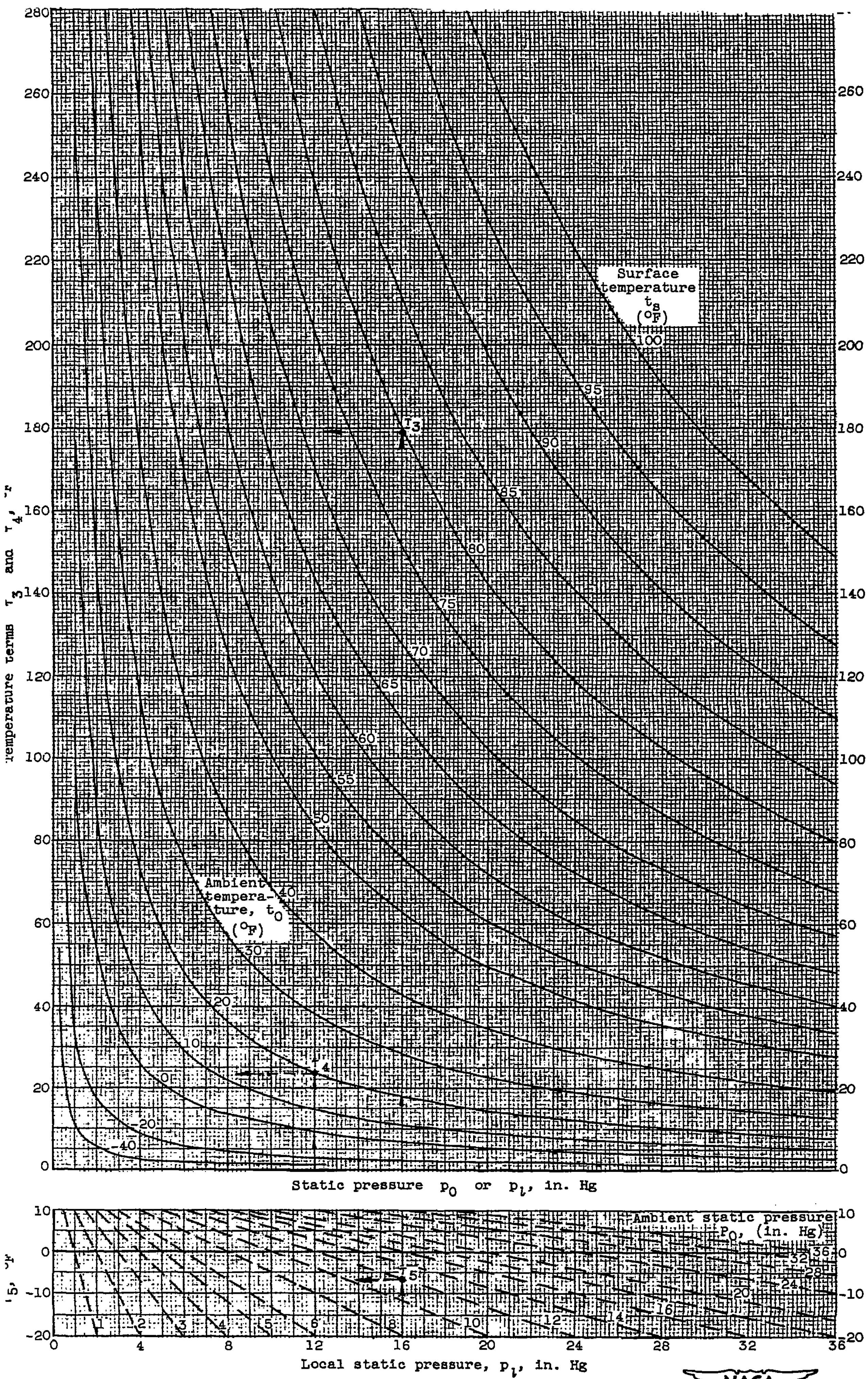


Figure 1. - Graphical solution of anti-icing heat and mass transfe



rom surface subject to impingement and heated above freezing.